Article

Design of an Anti-Slip Mechanism for Wheels of Step Climbing Robots

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Abstract: This manuscript presents a shape memory alloy (SMA) actuated anti-slip mechanism for the wheels of step climbing robots. The proposed mechanism comprises three kinematic chains considering the *Lazy Tong* and the bi-stable four-bar mechanism. Chain 1 of the mechanism is used to clamp on the edges of the stairs to avoid slipping. The second chain of the mechanism is used to switch the mechanism between two stable positions, i.e., open position and closed position, of chain 1. For activating the mechanism, the third chain is employed which is based on SMA wire. Furthermore, the mechanism is designed to achieve passive switching from the open position to the closed position. Equations are developed to determine the dimensions of various members. Using those parameters, a 3D model of the proposed mechanism is developed. Stress analysis is performed and the model is found to be safe under a load of 250 N with a factor of safety of 3.025. The mechanism is attached to either side of a wheel of the outer radius of 290 mm. To analyze the kinematics of the mechanism, a three-dimensional model in MSC Adams is developed and studied. The force required by SMA actuator is found to be less than 5 N. The proposed mechanism may be used for various unmanned robotic systems while mitigating step-like obstacles in the path.

Keywords: mechanism; assistive technologies; actuators; smart materials; bi-stable mechanism; step-climbing robot; four-bar mechanism

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1. Introduction

Rapid developments in the field of robotics and automation have led to the use of robots along with humans for performing complex tasks. This has improved the accuracy and efficiency in performing various tasks. In some of the fields like surveillance, exploration, and rescue in dangerous locations (e.g., nuclear or fire disasters), robots are deployed instead of human rescuers. This significantly reduces the risk of the adverse effect on human rescuers, which may have been caused by the hazardous environments of the rescue location. In such rescue robots, one of the key features is to overcome obstacles or to be able to maneuver over steps of stairs.

Many researchers have been working rigorously and successfully in developing robots which can climb up/down the stairs. Such robots may be widely classified in three groups i.e., tracked, articulated leg, and hybrid system. Robots using a track belt as a driving mechanism are known as a track-based stair climbing robot [1–3]. Although tracked based systems are successful in climbing up/down the stairs, these robots are slower on the plane surface as compared to wheeled robots. In addition, some researchers have developed systems based on articulated legs [4]. The biped robot legs are based on the Stewart Platform. Each leg of the robot is controlled by six linear actuators. These actuators are connected to the waist of the robot by six universal joints (degree of freedom (DOF) of universal joint is 2). The actuators are connected to the foot by three spherical joints (DOF

= 3). Complex control design algorithms are used to actuate twelve actuators to maneuver such robots. These robots can overcome obstacles efficiently; however, complexity in design and control result in an increase in the development cost of such robots. In addition, the operating speed of such robots is slow in planes. Hence, to overcome such problems of robots on a plane and achieve step climbing, researchers have worked towards the development of hybrid systems. Such hybrid systems may be further categorized into two groups i.e., wheels with track and wheels along with articulated legs [5–16]. These hybrid robots are equipped with two systems, one for maneuvering on a plane surface and the other for overcoming obstacles. Depending upon the situation, either one of the two systems is activated. Researchers have worked on modular-based stair climbing robots [17]. In such robots, detachable modules are attached to the parent robots depending on the situation. Linear actuators are used in such modules to push the robot upward on the stairs. These robots require the presence of an operator for connecting the modules at the time of operation. Robots based on power linkage are developed to achieve step climbing [18,19]. In this system, clusters of wheels are added along with powered links. Wheel clusters are used to overcome obstacles, whereas powered links are used to push the system forward on the stairs.

An interesting hybrid mechanism based on the sliders is discussed in a series of articles [20–24]. Clamping mechanisms are used to clamp on the edge of steps and, with the help of sliders, the robot pushes itself forward on the steps. Some of the developed stair climbing robots are based on wheel clusters [25–34]. These wheel clusters usually consist of two or three wheels. To overcome stairs, these clusters of wheels rotate about its center/centroid of the line/triangle joining center to the wheels. Some researchers used rotating multi-limb structure to maneuver the robot on stairs [35,36]. Another research work based on the slider-crank mechanism is developed [37–39]. In this robot, manipulating the center distance between the front and rear wheels by means of a slider–crank mechanism, step climbing is achieved. We may notice that the track-based and articulated step climbing robots are relatively slower on the in-plane surface, whereas in hybrid robots, due to use of two different mechanisms for plane motion and stair climbing, the weight and control complexity increase. Hence, among all the methods discussed, wheel based systems are better in terms of speed and simplicity of control while maneuvering on the plane as well as stairs. However, in the wheel based system, the successful operation of the system to climb up or down the stairs depends entirely on the friction between the wheel and the surface of stairs. If the friction between the wheel and surface of the stair is significantly low, the risk of slipping while climbing of the stairs is inevitable. Hence, a mechanism that can clamp on the edges of the stairs while climbing up or down is necessary to eliminate the dependency of the step climbing robot on the friction between stair surface and the wheels to maneuver on stairs.

In this article, we have proposed an array of SMA based clamping system attached to the wheels for step climbing. Clamping on the edges of steps will eliminate the possibility of the slipping of wheels on steps during climbing up or down the stairs. The proposed anti-slip wheel mechanism is comprised of three kinematic chains. These three chains act in conjunction to achieve successful clamping on stairs during climbing up or down the stairs. Each of the chains has a specific role in the proposed mechanism. Chain 1, also called the clamping mechanism, is responsible for clamping on the edge of the stairs. This chain has two operating positions, i.e., open position and closed position. The open position is activated during the motion of the climbing robot on the stairs. On the planar surface, the chain remains in a closed position. The shift between the closed position to the open position is achieved by an actuator mechanism, whereas the transition between the closed position to open position is achieved passively. Chain 2 of the mechanism is based on a bi-stable four-bar mechanism [40]. The two stable states of this chain help the kinematic chain 1 to be in either the open position or the closed position. In addition, this helps in reducing the length of actuation of the actuator by half. The third chain, Chain 3 (actuator mechanism), is based on a *Lazy-Tong* mechanism [41] and actuated by a SMA

based actuator. SMA based actuators are simple, lightweight, and power-efficient. Hence, SMA based actuators are chosen for actuation of the proposed anti-slip wheel mechanism. Kalra et al. have elaborately discussed the formulation of SMA wire based on a 1D Brinson model [42].

This paper is organized as follows: Section 2 explains about the design approach and the working principle; in Section 3, designing and modeling of the mechanism is discussed. Section 4 illustrates the results of the simulations. Section 5 concludes the paper.

2. Working Principle and Design Approach

In this section, the working principle and the design approach to select the proposed mechanism configuration of the proposed mechanism is explained.

2.1. Working Principle

Figure 1 shows the positions of the anti-slip mechanism during plane motion. It may be noticed that the proposed mechanism has two stable positions. These two positions are called the open position and closed position. The mechanism will remain in a closed position during plane motion as shown in Figure 1. While climbing up or down the stairs, the open position is activated using an SMA actuator, as shown in Figure 2a,b, respectively. While climbing up or down, the mechanism will clamp on the edges of the stairs to avoid any slipping of the wheels. This will result in improvement in the efficiency and safety of the climbing robot.

Figure 1. Working principle of an anti-slip mechanism on the plane surface.

Figure 2. Working principle of an anti-slip mechanism (a) while climbing on stairs; (b) while climbing down stairs.

In Figure 3a, a detailed view of the proposed mechanism is shown. The mechanism consists of three sets of kinematic chains, chains 1, 2, and 3, represented in green, red, and black colors, respectively. Encircled numbers attached to the links are used to denote different links and, throughout the article, these numbers are used to refer the links. Chains 1 and 3 are based on a *Lazy Tong* mechanism, and chain 2 is based on a bi-stable four-bar mechanism. Switching from the closed to the open position of the mechanism is done by passing a current through SMA wire, which increases the temperature of the wire because of joule heating. This results in shrinkage of SMA wire which in turn pushes the joint P_3 . Joint P_3 is a common joint of both chains 1 and 2 and connects link 1 and 4. In addition, the dotted red line on link 1 represents the shared part of the link by both chains 1 and 2. The bi-stable nature of the chain 2 is achieved by using torsion spring 2, which forces the joint P_3 to remain in either of the two extreme positions. The two extreme positions of the mechanism are limited by the slot L_{S1} in chain 1 (shown in green). The torsion springs in chain 1 are responsible for aligning of link 2 with link 1 when the mechanism clamps on the edges of steps. Alignment of links results in maintaining the open position of the mechanism throughout the entire rolling of the wheel on steps.

Figure 3. Detailed front view of the anti-slip mechanism (a) open position; (b) closed position.

Figure 3b shows the closed position of the proposed mechanism. Switching from the open to the closed position is achieved by the push generated by contact force between link 2 and the horizontal ground. Once all of the climbing up or down the stairs is completed, the supply of current to the SMA wire is stopped. This will result in a decrease in the temperature of the SMA wire, and the actuator will return to the elongated state. This will allow the joint P_3 to retract to the closed position of the mechanism when joint P_3 is pushed by the contact force between link 2 and the ground. However, while climbing up or down stairs, the current will continue to flow through the SMA wire. Hence, even if, due to any contact force between link 2 and the ground, the joint P_3 retracts, the linear spring will push joint P_3 back to the open position once the contact force disappears. This will allow the mechanism to be ready for clamping on upcoming steps.

2.2. Design Approach

In the previous section, working principle of the proposed mechanism is explained. In this section, the design approach to decide the configuration of the mechanism is illustrated. It may be observed from Figure 3 that link 2 and 2(a) of Chain 1 are used to clamp on the edges of a stair while climbing up and down the stairs, respectively. Use of a lazy tong mechanism to operate these two links helps to open the links with a single actuator instead of using one actuator each for link 2 and 2(a). Hence, this reduces the number of actuators. Using a bi-stable four bar mechanism in chain 2 helps in isolating the clamping mechanism from the actuating mechanism in the open position of the mechanism. Moreover, it helps in reducing the actuation length of the SMA actuator. The mechanism of chain 3 is used to utilize the property of an SMA actuator which shrinks when heat is applied to the actuator.

3. Design and Modeling

This section illustrates the design approach for the effective working of the anti-slip mechanism. Figure 4 represents a schematic of the proposed mechanism. Link numbers may be referred from Figure 3 (encircled numbers). The three chains of the proposed mechanism act in conjunction to successfully clamp on the edges of the stairs. Design approaches to determine the dimension of various members of the mechanism are explained in the following sections.

Figure 4. Schematic view of the anti-slip mechanism.

It may be noticed that, in a proposed mechanism, link 1 and link 2 are not parallel. Figure 6 illustrates the forces when link 2 touches the horizontal plane. Figure 6a represents the case when link 1 and link 2 are not parallel, whereas Figure 6b represents the case when link 1 and link 2 are parallel. The moment at point P_0 due to the resultant forces are given by:

$$M_{P_0} = f d_{P_0 P_2} \sin(\gamma - \angle P_0 P_2 P_1) \quad (2)$$

In Equation (2), γ for the case when link 1 is parallel to link 2, as shown in Figure 6b is negative and for the case when link 1 is parallel to link 2 are not parallel, as shown in Figure 6a γ is positive. In the first case, the resultant moment (M_{P_0}), about point P_0 , is in a counter-clockwise direction if the $\angle P_0 P_2 P_1$ is smaller than γ . However, the resultant moment (M_{P_0}) in the second case is in a clockwise direction. The resultant counter-clockwise moment in the first case will force link 2 to rotate in a counter-clockwise direction resulting in switching the mechanism to a closed position from the open position. This has two major advantages over the second case—first, saving the energy required by passively switching the mechanism from open to closed position. Second, avoid jerks on the wheel as in the second case the wheel will roll over the tip of the link 2.

Figure 6. Force equilibrium diagram when link 2 touches the horizontal plane (a) link 1 and link 2 are not parallel; (b) link 1 and link 2 are parallel.

Figure 7 shows the situation where the anti-slip mechanism is in the open position and the wheels are ascending the stairs. Figure 7a represents the mechanism having a fixed angle between link 1 and link 2, whereas, in Figure 7b, the joint between link 1 and link 2 is equipped with a torsional spring. This allows link 2 to rotate with respect to link 1 until link 2 is parallel to link 1. Such flexibility is an essential feature for the mechanism to function effectively. It may be observed from the first case that the net moment about point P_0 due to the reaction force at the point of contact between link 1 with the surface of step is in a counter-clockwise direction. Hence, the mechanism will switch to a closed position before the wheel touches the edge of the subsequent step. This may lead to a slipping of the wheel, jeopardizing the ascent of the robot. However, in the second case, the net moment due to the contact force will be in a clockwise direction. Since the links are designed in a way that the maximum angle between link 1 and link 2 is limited to 180° , the mechanism will remain active until climbing on a subsequent step is initiated.

Figure 7. Schematic view showing the section of a wheel while ascending the stairs (a) without torsion spring at joint P_1 ; (b) with torsion spring at joint P_1 .

Figure 8a represents a schematic view of half of the clamping mechanism as the other half of the mechanism operates symmetrically. In this section, kinematics of chain 1 are discussed. For a given distance between point P_0 and the center of the wheel 'O', the locus of the point P_2 with respect to the position of point P_4 is traced. Using these equations, the position and length (L_{S1}) of the slot are determined. This ensures the successful operation of the clamping mechanism. From the figure, it may be noticed that locus of point P_2 intersects two circles at points 'A' and 'B'. These circles are at an offset of $L_w/2$ (L_w is width of link 2) from the perimeter of tire and outer perimeter of the base plate, respectively. Figure 8b shows the detailed view of point 'A'. It may be noticed that the locus of point P_2 intersects a circle at an offset of $L_w/2$ from the perimeter of the tire. This is because of the width of link 2. Equations to express the maximum and minimum angle for link 1 are mentioned below.

Figure 8. (a) Schematic view of chain 1; (b) detailed view of point A.

$$d_{P_0P_2} = \sqrt{d_{P_2I}^2 + (d_{P_2D} + d_{HP_0})^2} \quad (3)$$

$$R_T = \sqrt{d_{P_2I}^2 + (d_{HP_0} + d_{P_0O})^2} \quad [R_D = R_T] \quad (4)$$

Subtracting Equation (4) from Equation (3);

$$d_{HP_0} = \frac{d_{P_0P_2}^2 - R_T^2 - d_{P_2D}^2 + d_{P_0O}^2}{2(d_{P_2D} - d_{P_0O})} \quad (5)$$

$$\begin{aligned} \alpha_{P_2} &= \cos^{-1} \left(\frac{d_{P_2D} + d_{HP_0}}{d_{P_0P_2}} \right) \\ \theta_1^{P_2} &= 90^\circ + (\alpha_{P_2} - \angle P_1P_0P_2) \end{aligned} \quad (6)$$

$$d_{P_0P_4}^{P_2} = d_{P_0P_3} \cos(\theta_1^{P_2} - 90^\circ) + \sqrt{(d_{P_0P_3} \sin(\theta_1^{P_2} - 90^\circ))^2 + d_{P_3P_4}^2} \quad (7)$$

where α_i is the acute angle made by point i with the vertical and vertex at P_0 . θ_1^k is the angle made by link 1 with horizontal in a counterclockwise direction and $d_{P_0P_4}^k$ is the distance between point P_0 and P_4 , when the tip of link 2 is at point k .

For a given d_{P_0O} (distance between point P_0 and center of wheel) and d_{P_2D} (distance between point P_2 and periphery of the tire), θ_1 and $d_{P_0P_4}$ are calculated. Similarly, a formula to obtain $d_{P_0P_4}^A$ is given by:

$$\begin{aligned} d_{GP_0} &= \frac{(R_T - L_w/2)^2 - d_{P_0P_2}^2 - d_{P_0O}^2}{2d_{P_0O}} \\ \alpha_A &= \cos^{-1} \left(\frac{d_{GP_0}}{d_{P_0P_2}} \right) \\ \theta_1^A &= 90^\circ + (\alpha_A - \angle P_1P_0A) \\ d_{P_0P_4}^A &= d_{P_0P_3} \cos(\theta_1^A - 90^\circ) + \sqrt{(d_{P_0P_3} \sin(\theta_1^A - 90^\circ))^2 + d_{P_3P_4}^2} \end{aligned} \quad (8)$$

A formula for $d_{P_0P_4}^B$ is given by:

$$\begin{aligned} d_{EP_0} &= \frac{(R_{RO} - L_w/2)^2 - d_{P_0P_2}^2 - d_{P_0O}^2}{2d_{P_0O}} \\ \alpha_B &= \cos^{-1} \left(\frac{d_{EP_0}}{d_{P_0P_2}} \right) \\ \theta_1^B &= 90^\circ + (\alpha_B - \angle P_1P_0A) \\ d_{P_0P_4}^B &= d_{P_0P_3} \cos(\theta_1^B - 90^\circ) + \sqrt{(d_{P_0P_3} \sin(\theta_1^B - 90^\circ))^2 + d_{P_3P_4}^2} \end{aligned} \quad (9)$$

subtracting Equation (9) from Equation (7), length L_{S1} may be determined.

The number of anti-slip mechanisms on a wheel depends on the outer diameter of the rim of a wheel. All the mechanisms are mounted on a plate, called the base plate, and then attached to the wheel. The outer diameter of the base plate will be equal to the outer diameter of the rim of the wheel. A minimum number of mechanisms required is given by Equation (10):

$$N = \frac{360^\circ}{\alpha_B} \quad (10)$$

The number of mechanisms attached to the base plate is the nearest integer value greater than N . Since two such base plates are attached to either side of the wheel, the number of the mechanism attached to a wheel is twice as in a single base plate.

3.2. Chain 2: Bi-Stable Four Bar Mechanism

The two positions of the anti-slip wheel mechanism, i.e., open and closed position, are achieved by using a bi-stable four-bar mechanism. A four-bar mechanism where a torsion spring is attached to one of the joints resulting in two stable positions is known as a bi-stable four-bar mechanism. In Figure 9a, a schematic view of the proposed bi-stable mechanism is shown. A torsion spring is attached at a joint P_6 . Two stable positions of link 1 are shown in Figure 9b by the line joining point P_0 and P_3 , and P_0 and P'_3 . In addition, a critical position is shown in the figure by a line joining P_0 and P'_3 . At this position, link 1 and link 4 are parallel and the mechanism may be stuck in this position. Hence, for the successful operation of the proposed mechanism design, parameters are chosen to avoid the critical position.

Figure 9. (a) Schematic view of chain 2; (b) two stable positions on bi-stable mechanism.

$$\begin{aligned}
 d_{P_0P_6} &= \sqrt{d_{P_0O}^2 + d_{OP_6}^2 - 2d_{P_0O}d_{OP_6}\cos\beta_6} \\
 \theta_3 &= \theta_1 + 180^\circ \\
 d_{P_3P_6} &= \sqrt{d_{P_0P_3}^2 + d_{P_0P_6}^2 - 2d_{P_0P_3}d_{P_0P_6}\cos(\theta_{P_6} - \theta_3)} \\
 \angle P_0P_6P_3 &= \cos^{-1}\left(\frac{d_{P_0P_6}^2 + d_{P_3P_6}^2 - d_{P_0P_3}^2}{2d_{P_0P_6}d_{P_3P_6}}\right) \\
 \angle P_3P_6P_5 &= \cos^{-1}\left(\frac{d_{P_3P_6}^2 + d_{P_5P_6}^2 - d_{P_3P_5}^2}{2d_{P_3P_6}d_{P_5P_6}}\right) \\
 \theta_6 &= \theta_{P_6} + \angle P_0P_6P_3 + \angle P_3P_6P_5 - 360^\circ \quad [For \theta_3 \leq \theta_{P_6}] \quad (11) \\
 \theta_6 &= \theta_{P_6} - \angle P_0P_6P_3 + \angle P_3P_6P_5 - 360^\circ \quad [For \theta_3 > \theta_{P_6}] \quad (12)
 \end{aligned}$$

Critical angle for link 1 is given by following equation:-

$$\theta_{3CR} = \cos^{-1}\left(\frac{(d_{P_0P'_3} + d_{P'_3P'_5})^2 + d_{P_0P_6}^2 - d_{P'_5P_6}^2}{2(d_{P_0P'_3} + d_{P'_3P'_5})d_{P_0P_6}}\right) \quad (13)$$

The link dimensions must be chosen in a way that, during passive switching, the mechanism from the open to the closed position the angle θ_3 must be more than θ_{3CR} , whereas, during active switching from a closed position to an open position, angle θ_3 must be less than θ_{3CR} .

3.3. Chain 3: Actuator Mechanism

To actuate the mechanism, an actuator based on a shape memory alloy (SMA) is used. Figure 10a represents the schematic of the actuation mechanism. An SMA wire is installed between point P_7 and point P_{10} . When a current is passed through the SMA wire, due to the electrical resistance of the wire, the temperature of the wire rises. This results in shrinkage of the wire. The points P_8 , P_{11} , and P_3 are constrained to move along the link 6 with the help of a slot. A linear spring is attached between point P_8 and P_{11} . As the SMA wire shrinks, the distance between point P_7 and P_8 increases, pushing the point P_3 to switch the mechanism from a closed to an open position.

Figure 10. (a) Schematic view of chain 3; (b) situation when the ground force pushes the mechanism to a closed position but the actuator in an open position.

It may be noticed that point P_3 represents the joint between link 1 and link 3, whereas P_{11} represents the endpoint of linear spring. In the closed position, these two points will be in the same position due to the force exerted by the linear spring. However, during the open position of the mechanism, the points will be in separate positions. Figure 10b illustrates a situation when the ground force pushes the anti-slip mechanism to a closed position, but the SMA actuator is powered on. Such situation may arise when the robot is on an intermediate platform while climbing on the stairs. It may be noticed from the figure that, although the ground force pushes the mechanism to closed position, since the actuator is powered on, a deflection in link 2 results in compression of a linear spring. The energy stored during compression of a linear spring will push the linkage to an open position as soon as the ground contact force vanishes. Use of a linear spring in between the actuation mechanism and clamping mechanism has two benefits. Firstly, it helps in reducing the impact on the actuator. Secondly, any force pushing the mechanism to a closed position, when the actuator is powered on, it pushes the mechanism to an open position upon withdrawal of the force. This results in keeping the mechanism open for any subsequent climbing:

$$\begin{aligned}
 d_{P_7P_3} &= \sqrt{(P_{7x} - P_{3x})^2 + (P_{7y} - P_{3y})^2} \\
 d_{P_7P_8} &= d_{P_7P_3} - d_{P_8P_{11}} \quad [d_{P_8P_{11}} = \text{Length of linear spring}] \\
 \angle P_7P_9P_8 &= \cos^{-1} \left(\frac{d_{P_7P_9}^2 + d_{P_8P_9}^2 - d_{P_7P_8}^2}{2d_{P_7P_9}d_{P_8P_9}} \right) \\
 \angle P_7P_9P_{10} &= 180^\circ - \angle P_7P_9P_8 \\
 d_{P_7P_{10}} &= \sqrt{d_{P_7P_9}^2 + d_{P_9P_{10}}^2 - 2d_{P_7P_9}d_{P_9P_{10}}\cos(\angle P_7P_9P_{10})} \quad (14)
 \end{aligned}$$

where P_{i_x} is the projection point P_i on the x -axis and P_{i_y} is the projection point P_i on the y -axis (with origin of the co-ordinate system at the center of the wheel). Equation (14) is used to determine the distance between point P_7 and P_{10} at different stages of the mechanism.

3.4. Force Requirement of the SMA Actuator

In this section, the force required achieving the variation in the actuator length is discussed. Figure 11 represents the schematic of chain 2 and chain 3. It may be noticed that the SMA wire must generate enough force to overcome the torque generated by the torsional spring 2 while rotating link 5 from point P_5 to point P'_5 . Point P'_5 represents the case when the bi-stable four-bar mechanism is at a critical state.

Figure 11. Schematic view of chain 3 and chain 2 for SMA force calculation.

The relationship to derive the maximum force required by the actuator for the successful operation of the mechanism is based on the virtual work method. The maximum force required to push point P_3 beyond will be at the critical position of the bi-stable mechanism as the resistivity torque by the torsion spring 2 will be the maximum at this position:

$$\begin{aligned} F\delta S &= T\delta\theta \\ F &= \tau\Delta\theta \frac{\delta\theta}{\delta S} \quad [T = \tau\Delta\theta] \end{aligned} \quad (15)$$

where δS is the infinitesimal change in distance between point P_7 and P_{10} . T is the torque generated by the torsional spring 2, and $\delta\theta$ is the infinitesimal change in angle θ_6 with respect to δS . τ is the stiffness of the torsional spring. $\Delta\theta$ is the difference in the angle made by link 5 with a horizontal at the closed position and the critical position:

$$\begin{aligned} d_{P_0P_7} &= \sqrt{2R_{P_0}^2(1 - \cos\beta_7)} \\ \angle P_7P_0P_3 &= \cos^{-1}\left(\frac{d_{P_0P_3}^2 + d_{P_0P_7}^2 - d_{P_3P_7}^2}{2d_{P_0P_3}d_{P_0P_7}}\right) \\ \theta_3 &= \theta_{P_0P_7} - \angle P_7P_0P_3 \end{aligned} \quad (16)$$

where $\theta_{P_0P_7}$ is the angle made by line joining point P_0 and point P_7 with the horizontal in a clockwise direction.

Using Equations (13), (14) and (16) relation between $d_{P_7P_{10}}$ and θ_6 may be derived at a critical position. Differentiating the relation, the ratio of $\delta\theta$ to δS may be derived and given as:

$$\frac{\delta\theta}{\delta S} = \frac{-d_{P_7P_{10}}d_{P_8P_9}d_{P_7P_{11}}d_{P_0P'_5}\sin\angle P_7P_9P_8\cos\angle P'_5P_0P_6}{d_{P_9P_{10}}d_{P_7P_8}d_{P_0P_7}d_{P_0P_3}d_{P'_5P_6}\sin\angle P_7P_9P_{10}\sin\angle P_7P_0P_{11}\cos\angle P_0P_6P'_5} \quad (17)$$

3.5. Actuation Length of an SMA Actuator

Actuation length of an actuator is one of the important factors in the designing of the actuator. In Section 3.4, the force required to actuate the mechanism from the closed position to overcome the critical position is discussed. Using this force, the required change in the length of SMA actuator (ΔX) may be determined. The change in actuator length comprises two components, i.e., change is actuator length due to a change in the length of linear spring (ΔX_L) and change is actuator length due to a deflection of link 5 (ΔX_θ). The equation of actuation length is given as:

$$\Delta X = \Delta X_L + \Delta X_\theta \quad (18)$$

For an applied force (F) by the SMA actuator which causes deflection of link 5 by $\Delta\theta$, the change in actuator length is given by Equation (15):

$$\Delta X_\theta = \frac{\tau\Delta\theta^2}{F} \quad (19)$$

Change is length of linear spring due to applied force by the actuator being caused by the axial force (F') on link 6 at point P_8 :

$$\begin{aligned} \Delta X_L &= \frac{F'}{K} \\ \Delta X_L &= \frac{-Fd_{P_7P_8}d_{P_9P_{10}}\sin\angle P_7P_9P_{10}}{Kd_{P_8P_9}d_{P_7P_{10}}\sin\angle P_7P_9P_8} \end{aligned} \quad (20)$$

Using Equations (19) and 20 in Equation (18), ΔX may be determined.

4. Results and Discussion

An anti-slip wheel mechanism is designed using the equations discussed in Section 3 for a wheel of outer diameter 580mm. Dimensions of different members are chosen for the successful operation of the proposed mechanism. The dimensions of the members are shown in Table 1.

Table 1. Dimensions of the members of the anti-slip mechanism.

Parameters	Values	Parameters	Values
Link 1	35 mm	R_T	290 mm
Link 2	50 mm	R_{RO}	275 mm
Link 3	35 mm	R_{RI}	185 mm
Link 4	25 mm	R_{P_0}	255 mm
Link 5	50 mm	R_{P_6}	230 mm
Link 6	42 mm	L_{S1}	18 mm
Link 7	12 mm	β_6	10°
Link 8	30 mm	β_7	9°
Spring length	10 mm	α	140°

It is found that 13 such mechanisms should be attached to one side of the wheel. To minimize the load distribution at the tip of link 2, another set of the proposed mechanism is attached to the other side of the wheel. Hence, 26 mechanisms (13 on either side of

the wheel) with the dimensions mentioned in Table 1 should be attached to the wheel of diameter 580 mm.

4.1. Stress Analysis

The developed 3D model is used to perform stress analysis. Figures 12 and 13 represent different views of the developed 3D model. Figure 12a shows the front view of the model when the mechanism is in an open position. It may be noticed that the tip of link 2 is protruded outside the outer diameter of the wheel. This will allow the mechanism to clamp on the edge of the steps. Figure 12b shows the side view of the 3D model. This figure illustrates the mounting of two anti-slip mechanisms on either side of the wheel. This results in sharing the load on the individual wheel. Figure 13 shows a perspective view of the developed model.

Figure 12. 3D model of anti-slip wheel mechanism (a) front view; (b) side view.

Figure 13. Perspective view of the 3D model of the anti-slip wheel mechanism.

Stress analysis is performed on chain 1 of the mechanism, chain 1 experiences the maximum external loads during its engagement with the edge of the stairs. Links' materials are selected as ASTM A36 (structural grade material), having a yield strength of $6.204 \times 10^8 \text{ N/m}^2$. A load of 250 N is applied and fixing the base plate as a constraint, Von-Mises stress is evaluated. Table 2 shows the details of finite element analysis. Figures 14 and 15 show different views of the contour plot of Von-Mises stress on different members of chain 1. In Figure 14, location and direction of applied load are displayed. In Figure 15, the point at which maximum Von-Mises stress developed is shown. The max-

imum of stresses developed in various members of the chain 1 is $2.051 \times 10^8 \text{ N/m}^2$ and the factor of safety is 3.025. This validated the safety of link members under the applied load of 250 N. Figure 16 represents the deflection on the members under the applied load. The maximum displacement of 0.205 mm is observed at the tip of link 2.

Figure 14. Von-Mises stress on chain 1 due to external loading (View 1).

Figure 15. Von-Mises stress on chain 1 due to external loading (View 2).

Figure 16. Deflection in members of chain 1 due to external loading.

Table 2. Details of finite element analysis.

Mesh Details	Values
Mesh type	Solid mesh (Tetrahedral)
Mesher used	Standard mesh
Jacobian points	4 points
Element size	8.92029 mm
Tolerance	0.446014 mm
Adaptive Method	None

4.2. Motion Analysis

To analyze the motion of the proposed mechanism, a 3D model (for link dimension, refer to Table 1) is developed in MSC-Adams. Figure 17a shows the closed position of the mechanism and Figure 17b shows the open position. It may be noticed that, in open position, links 2 and 2(a) have moved beyond the radius of the tire. This will ensure clamping by the mechanism on the edge of the stairs.

Figure 17. Model of proposed mechanism developed in MSC Adams (a) closed position; (b) open position.

For the analysis of the mechanism, the coefficient of stiffness of linear spring (between point P8 and P11 of actuator mechanism) is considered as 15 N/mm. The coefficient of stiffness and damping of torsional spring at point P6 are considered as 5 N-mm/deg and 1 N-mm-s/deg, respectively. The simulation is performed for 2 s with 5000 simulation steps. Figure 18a shows the variation of angle of link 1a, the angle between link 1 and link 4, and the angular deflection of link 5. It may be noticed that the angle between link 1 and link 4 ($\angle P5P3P0$) becomes 180° at 1.13 s. This implies that link 1 and link 4 are inline. Once the mechanism crosses this position, the mechanism moves to its next stable state, which results in the open position of the anti-slip mechanism. It may be observed that the angular displacement of link 5 (link 5 is attached to the torsional spring which forces the mechanism into bi-stable state) reaches 10° (maximum displacement) and remains unchanged until 1.4 s. This behavior is observed because of the damping of the torsional spring. After 1.4 s, the rate of change of angular displacement increases and the mechanism attains the open position of the anti-slip mechanism. This increase in angular velocity of link 1a and 5 may be observed in Figure 18b.

For actuating the anti-slip mechanism mechanism to the open position, an SMA wire is used (refer to Figure 3). Figure 19a illustrates the variation of force on the SMA wire during the opening of the proposed mechanism. It may be noticed that the maximum force required to be generated by the SMA actuator is 4.7 N. For changing the mechanism position from the closed to open position, the SMA wire should shrink by 7 mm, which may be observed from Figure 19b.

Figure 18. Kinematics of links 1a and 5 (a) angular displacement; (b) angular velocity.

Figure 19. (a) Force required by SMA actuator; (b) deformation of SMA actuator.

To validate the mechanism, simulations are performed by attaching a single pair of the proposed mechanism to a wheel of diameter 580 mm. The material for step and links are considered to be carbon steel (grade A36), whereas, for the tire, rubber is considered. Figures 20 and 21 shows snapshots of the simulations for ascending and descending, respectively. The zoomed in view of the contact area between link of mechanism and the surface of stair is also shown in the figures. From Figure 20a, it may be observed that

initially links 1 and 2 are at an angle, whereas, in Figure 20b–d, when link 2 comes in contact with the surface on the stairs, the angle between link 1 and link 2 becomes 180° . The links remain in-line until the angle between link 2 and the surface becomes greater than 90° , which may be observed from Figure 20e. Similarly, Figure 21 illustrates the simulation for descending of the wheel. In Figure 21a, it may be observed that the wheel is not in contact with the edge of the stair. In such condition, a wheel without an anti-slip mechanism will tend to slip off the edge. It may be observed that link 2 of the mechanism is in contact with the surface of the stair, and this helps to prevent the slipping of the wheel.

Figure 20. Motion simulation of the mechanism while ascending a step (time stamp in milliseconds are roughly estimated from videos) (a) the tip of link 2 is about to touch the edge of the step; (b) the tip of link 2 touches the edge of the step; (c) link 2 becomes parallel to link 1; (d) wheel climbs half way; (e) as the climbing is about to complete link 2 returns to initial position; (f) climbing is complete.

Figure 21. Motion simulation of the mechanism while descending a step (time stamp in milliseconds is roughly estimated from videos) (a) the wheel starts to roll with tip of link 2 as a pivot; (b) as the wheel rolls down, the tip of link 2 touches the edge of the step preventing the wheel from slipping; (c) descending of wheel is complete; (d) wheel moves forward in plane after completion of descending.

5. Conclusions

In this article, an anti-slip mechanism for the wheels of step climbing robots is designed. An array of proposed mechanisms is attached to the wheels of the robot to eliminate the possibility of slipping while maneuvering on stairs. The mechanism enables the robot to clamp on the edges of stairs to prevent the slipping while climbing stairs. This will make it capable of providing efficient and safe climbing operations. Furthermore, using the relationships obtained for the proposed mechanism, a 3D model of the mechanism is developed for a wheel of diameter 580 mm. The performance of the model is found to be satisfactory and verified through Von-Mises stress and deflection analysis under an applied load up to 250 N. The factor of safety in the stress analysis of the mechanism is found to be 3.025. The proposed mechanism may be attached to the wheels of wheel-based robots. This will reduce the risk of the slipping of wheels on the edges of a stair, which in turn improves the climbing capabilities of the robot on the stairs. Such robots may be used for the door-to-door delivery robots, where robots need to climb up/down the stairs of a building to deliver goods at the door. Another application of such a system is to use it in exploration or rescue robots where human deployment is a risky operation. In the future, optimization analysis may be performed to optimize various link parameters. A prototype will be developed and experiments will be carried out to ensure the safe operation of a step climbing robot while climbing up or down the stairs.

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Abbreviations

The following abbreviations are used in this manuscript:

R_T	Radius of tire
R_{RO}	Outer radius of base plate
R_{RI}	Inner radius of base plate
R_{P_x}	Radius of point P_x (x is variables)
L_{S1}	Length of slot 1
β_6	Angle of point P_6 with vertical
β_7	Angle of point P_7 with vertical
α	Angle between link 1 and link 2
$\angle ABC$	Acute angle between points A, B, and C with B as vertex (A,B,C are variables)
d_{ij}	distance between points i and j
θ_i	Angle made by link i with horizontal (in a counterclockwise direction)

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